Analysis of the impact of ignition timing and air-fuel ratio on indicated efficiency for biogas engines

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ABSTRACT

This article examines the impact of the air-fuel ratio, measured by lambda probe, and the ignition timing on the indicated efficiency of an internal combustion engine using biogas as fuel. Through analysis, it's seen that the indicated efficiency increases as the air-fuel ratio leans towards a stoichiometric value, and as the ignition timing advances towards the top dead center. The study presents analysis results in the form of graphs. The potential benefits of using biogas as a renewable energy source are highlighted, given its lower carbon footprint compared to fossil fuels. The findings of this article provide valuable insights for the development of efficient and sustainable engines powered by biogas.

INTRODUCTION

The major focus on emissions and efficiency in recent research on internal combustion engines is due to the ever-increasing legal requirements for emissions levels and the highly competitive automotive market, always searching for new solutions. The study of the impact on volumetric efficiency of alternative fuels is part of this context, as it directly affects the performance of several engine parameters, including efficiency and emission levels. The use of renewable fuels such as biogas generates various benefits beyond environmental ones, such as the utilization of byproducts and job creation.

Studies on modeling and control of internal combustion engines aim to develop the best strategy for the engine, ensuring that the states in which the fuel capacity is best utilized are employed. It begins with a model derived from the study of the engine's characteristics and behaviors, followed by the application of the best control strategy. The model should have characteristics such as being applicable in real-time and a viable approximation of the engine's actual behavior.

The study of the impact on engine efficiency caused by variations in the air-fuel ratio and ignition angle must be known to model the generation of energy and the efficiency of the internal combustion engine. A precise efficiency model that can be used in real-time is important for designing control strategies that balance performance and emission levels. Changes in the air-fuel ratio and ignition angle are among the main parameters used to estimate indicated efficiency.

The impacts on efficiency caused by variations in the air-fuel ratio and ignition angle will be demonstrated and equated in this article, allowing for the modeling of indicated efficiency. The article is organized into sections. Section 1 explains the methodology used. Section 3 presents the model of indicated efficiency and its dependence on the air-fuel ratio and ignition timing. Section 4 presents the experiments conducted and the experimental conclusions. Finally, acknowledgments and general conclusions are presented.

METHODOLOGY

A literature review is conducted on the impact of ignition angle and air-fuel ratio on efficiency to understand various approaches that analyze this relationship. Then, an experimental plan is developed to confront the theory studied with laboratory results. It is also ensured that the experimental setup has all the necessary aspects, such as appropriate sensors. Then, changes in ignition angle and air-fuel ratio values are made in the laboratory to assess the impact on indicated efficiency.

MODELING

In this section, we will model the average torque value produced in a cycle. For this, we will use two definitions. The first is the brake mean effective pressure, which is the pressure that must act on the piston during the expansion phase to produce the same amount of work that the real engine produces in a complete cycle (two engine revolutions, considering a four-stroke engine). The second is the fuel mean effective pressure, which is the brake mean effective pressure that an engine with 100% efficiency would produce with the injected fuel mass m_{φ} (i.e., with perfect conversion of the fuel's lower heating value into mechanical energy) [1].

Therefore, from the previous definitions, we can develop the following formulas for brake mean effective pressure and fuel mean effective pressure, respectively:

$$p_{m\varphi} = \frac{H_l \cdot m_{\varphi}}{V_d} \tag{1}$$

In which,
H_l: Lower heating value of the fuel;
T_e: Mean torque value.

The amount of fuel mass that enters the cylinder in one cycle can be calculated from the average value formulation of the mass flow of fuel admitted by the cylinder:

$$m_{\varphi} = \frac{4\pi}{\omega_e(t)} \cdot \dot{m_{\varphi}} \tag{2}$$

Therefore, we can describe the efficiency of an engine as:

$$\eta_e = \frac{p_{me}}{p_{m\varphi}} = \frac{T_e \cdot 4\pi}{m_{\varphi} \cdot H_l} \tag{3}$$

In this study, we will model the torque taking into account the average fuel flow entering the cylinder, the air/fuel ratio, the engine speed, the ignition angle, the recirculation rate, and the fuel composition. As there are phenomena that are not related to the mass of fuel injected in one cycle, and as the variation of efficiency with the mass of fuel injected is insignificant (except under high load conditions [1]), we will have as our first simplification:

$$p_{me} = e(\omega_e, \lambda, \zeta \ldots) \cdot p_{m\varphi} - p_{me0}(\omega_e, \vartheta_e, r_l, \ldots) \tag{4}$$

In which,
$e(.)$: Simplified η_e efficiency;
ζ: Ignition angle;
p_{me0}: Phenomena that are not related to the amount of fuel injected per cycle;
r_l: Relative charge.

By employing the optimum ignition angle for a given operating condition, we maximize the produced torque and the motor efficiency while keeping the fuel consumption minimum. The maximum torque achieved in this way is called the maximum brake torque. The optimum ignition angle for which the maximum brake torque occurs for each operating condition is obtained by varying the ignition angle while measuring the brake torque (output torque at the crankshaft) through a dynamometer. Traditionally, in modern vehicles, the values of the optimum ignition angle are stored in maps [2].

Modeling the optimum ignition angle is necessary because excursions from the optimum angle are one of the main causes of efficiency loss. Other major causes include the ideal maximum energy conversion limit for the Otto cycle, the thermodynamic processes and their respective relationship with the engine speed, and the oxygen deficit for rich mixtures.

The weighting of λ in the efficiency modeling aims to take into account aspects such as partial burns, excess fuel, and residual fuel content. Since these factors depend only on λ, we can make the following simplification:

$$e(\omega_e, \lambda, \zeta, x_{egr}, \Upsilon \ldots) = e(\omega_e, \zeta, \Upsilon \ldots) \cdot e_{\lambda}(\lambda, \Upsilon) \tag{5}$$

As we wish to take into account in the modeling of thermal efficiency losses in thermodynamic processes that occur even at the optimum ignition angle and that are related to the engine speed, such as heat transfer and combustion limited by the time available within a motor cycle, we will make the following decomposition, arriving at the final simplification as presented in (3.8):

$$e(\omega_e, \lambda, \zeta, x_{egr}, \Upsilon \ldots) = e_{\omega_e}(\omega_e, \Upsilon) \cdot e_{\lambda}(\lambda, \Upsilon) \cdot e_{\zeta}(\zeta, \Upsilon) \tag{6}$$

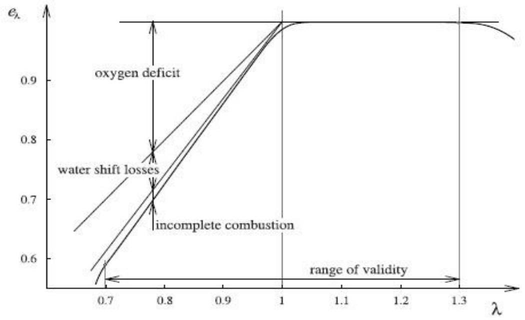
Changes in λ result in losses in efficiency due to various factors. In the operational range, to be identified for each engine, but usually between $0.7 \leq \lambda \leq 1.3$, the main factors that influence $e_{\lambda}(\lambda)$ are oxygen deficit, fuel-air ratio variation, and incomplete combustion. For $\lambda < 0.7$ or $\lambda > 1.3$, due to fuel excess or shortage, respectively, ignition failures occur [3]. Therefore, efficiency in this region should drop abruptly.

This analysis of losses due to RMV is important, as they are more representative than losses due to incomplete combustion [1]. However, we can find in the literature research where incomplete combustion and RMV were neglected, as in [4].

In [5], there is a proposal to not use a curve $e_{\lambda}(\lambda)$ to be stored in the simulation hardware memory, but to use a formulation for $e_{\lambda}(\lambda)$ as follows:

$$e_\lambda(\lambda) = \min(1, \lambda) \qquad (7)$$

Figure 1: Normalized curve of λ influence on indicated efficiency.



Source: [1]

The way in which the ignition-dependent efficiency factor varies as excursions occur from the optimal ignition angle for a single-fuel engine depends only on the mechanical design of the engine and not on the operating point [6]. John Moskwa showed in [7] that the $e_\zeta(\zeta)$ curve can be approximated by a parabola with a maximum value at the optimal ignition angle.

In [1], we find a formulation for the $e_\zeta(\zeta)$:

$$e_\zeta(\zeta) = 1 + k_\zeta \cdot \left(\zeta - \zeta_0(\omega_e, p_{adm})\right)^2 \qquad (8)$$

EXPERIMENTAL STRUCTURE

In this experiment, the engine was used coupled to a bench dynamometer to adjust the engine speed. The intake pressure, injection timing and ignition angle are controlled using the INCA software, and the data were recorded in Matlab software. A programmable ECU was used, which communicates with the INCA software. The engine used is the EA 111 VHT 1.6 liter, modified to work with indirect gas injection.

Table 1: Instrumentation used.

Pre-catalytic lambda sensor	LSU4.9
Passive bench dynamometer	Schenck Type D 360 1E hydraulic
Active bench dynamometer	Antriebstechnik INDY 33/4P
Lambda measurement analyzer	ETAS LA4
Analog input reader	ETAS ES650
Gasoline FLEX ECU	BOSCH MED17ETAS-2.41
Calibration and acquisition software	INCA v7.1
Engine	EA 111 VHT 1.6 liter

Figure 2: EA 111 VHT 1.6 liter engine coupled to the Schenck Type D 360 1E hydraulic Passive bench dynamometer.

Figure 3: Programmable ECU connected to a data acquisition board.

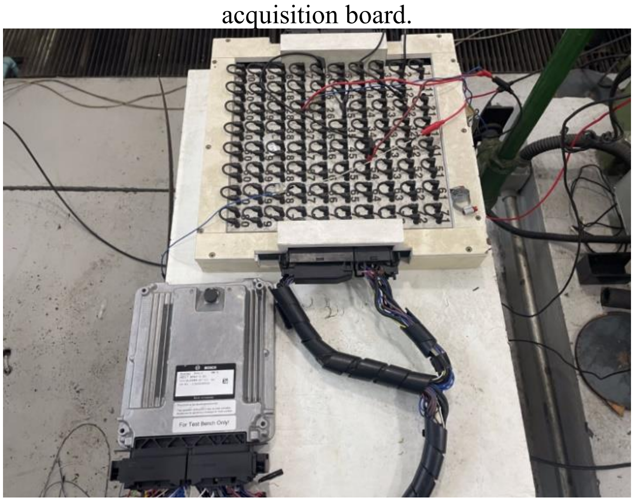
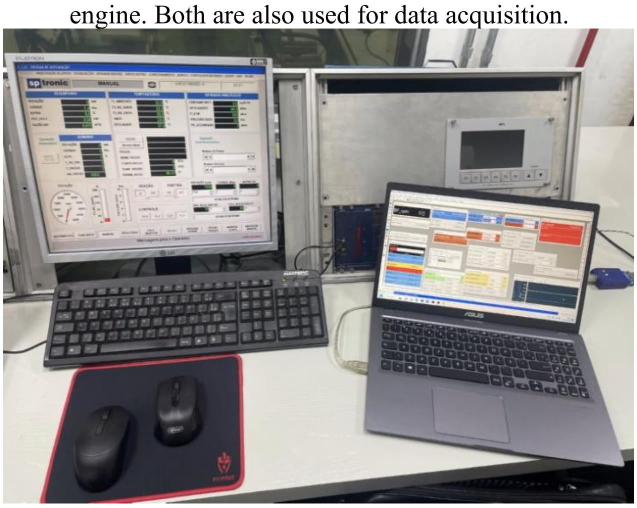


Figure 4: E_CAT software used to control the dynamometer, and INCA software used to control the engine. Both are also used for data acquisition.



RESULTS

Initially, in order to determine the impact of ignition timing on effective torque, the ignition timing was varied for different engine speeds and intake pressures, as shown in figures 5 to 8, and the torque magnitude was measured. The results obtained confirm that torque is a quadratic function of ignition timing. All the curves in figures 5 to 8 were plotted for stoichiometric air-fuel ratio.

Figure 5: Effective torque as function of ignition timing for different intake pressures and engine speed at 1500 RPM.

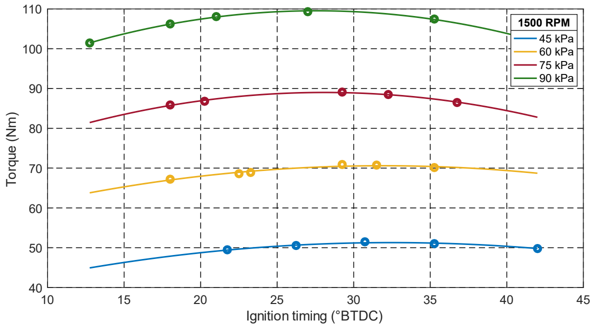


Figure 6: Effective torque as function of ignition timing for different intake pressures and engine speed at 2000 RPM.

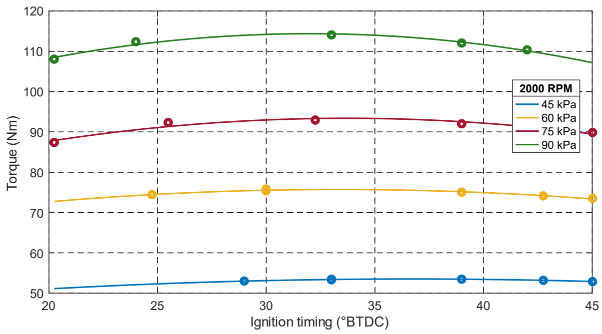


Figure 7: Effective torque as function of ignition timing for different intake pressures and engine speed at 3000 RPM.

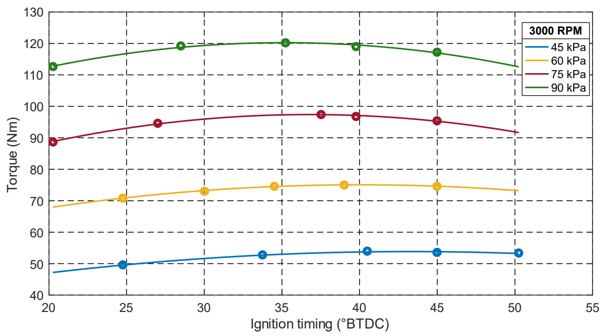
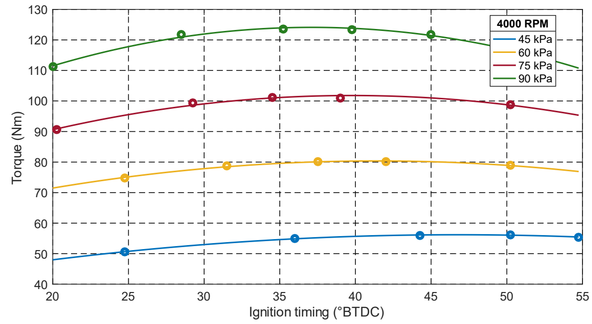
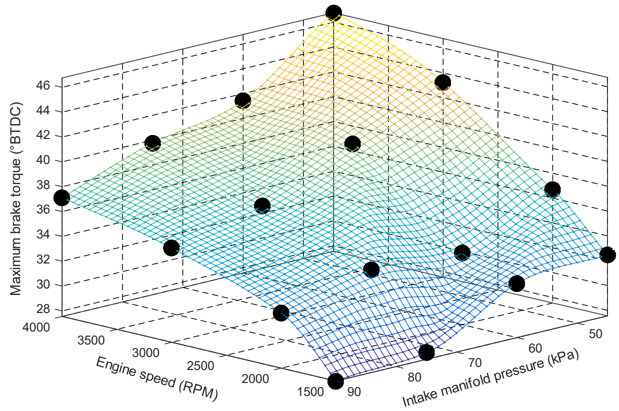


Figure 8: Effective torque as function of ignition timing for different intake pressures and engine speed at 4000 RPM.



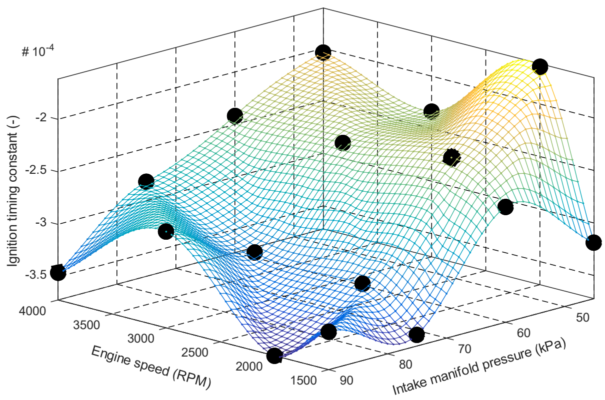
The optimal ignition timing, at which the torque is maximum for a given condition, is referred to as the maximum brake torque (MBT) timing. In Figure 9, all the optimal ignition timings were mapped as a function of engine speed and intake pressure. The value of the MBT angle corresponds to the variable $\zeta_0(\omega_e, p_{adm})$ from equation 8.

Figure 9: Maximum brake torque timing as function of engine speed and intake manifold pressure.



The parameter k_ζ from equation 8 is mapped as a function of engine speed and intake manifold pressure. For greater accuracy in the model, this dependence on engine speed and intake manifold pressure can be considered. However, k_ζ is widely treated as constant by the literature [1].

Figure 10: Ignition timing parameter k_ζ as function of engine speed and intake manifold pressure.



Different air-fuel ratios were used to understand the impact of lambda ratio variations on effective torque. Tests were performed for different engine speeds and intake pressures. The results obtained can be seen in figures 11 to 14. Next, the torque losses due to friction and gas pumping losses were added to the effective torque. By normalizing the torque values thus attained, the normalized combustion efficiency is obtained [5]. The results can be viewed in figures 15 to 18.

Figure 11: Effective torque as function of lambda ratio for different intake pressures and engine speed at 1500 RPM.

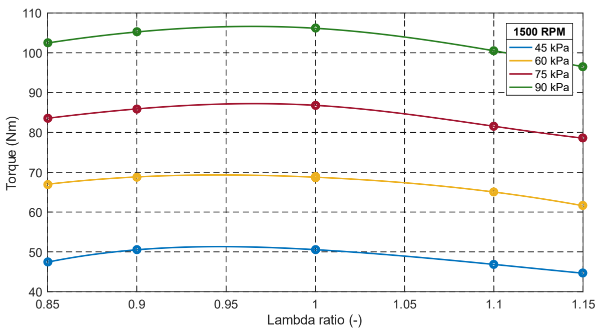


Figure 12: Effective torque as function of lambda ratio for different intake pressures and engine speed at 2000 RPM.

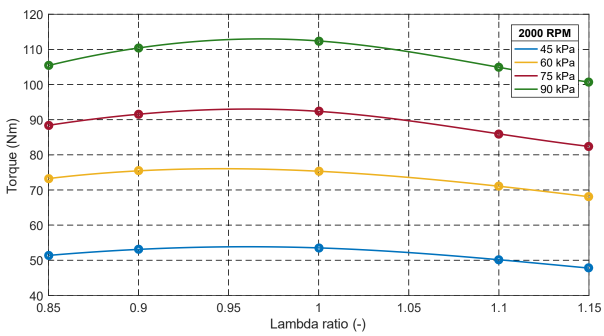


Figure 13: Effective torque as function of lambda ratio for different intake pressures and engine speed at 3000 RPM.

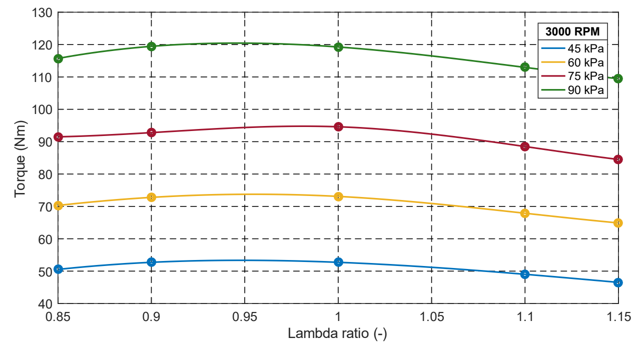


Figure 14: Effective torque as function of lambda ratio for different intake pressures and engine speed at 4000 RPM.

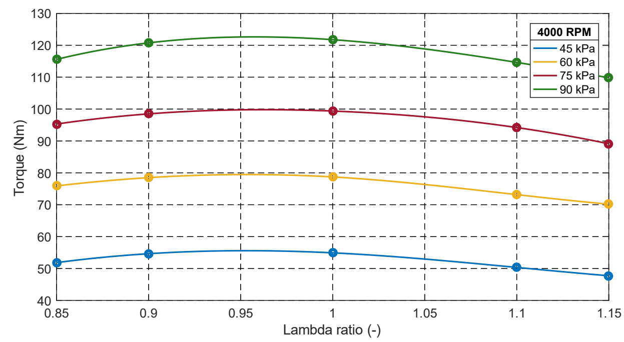


Figure 15: Normalized combustion efficiency as function of lambda ratio for different intake pressures and engine speed at 1500 RPM.

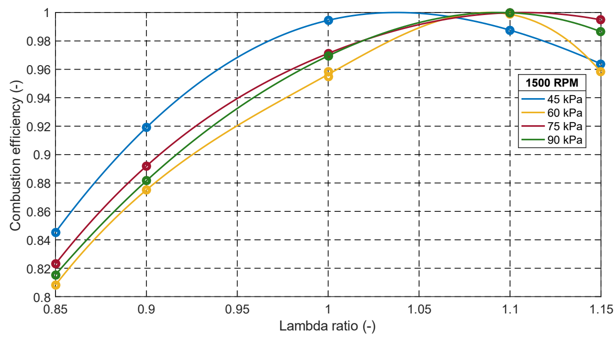


Figure 16: Normalized combustion efficiency as function of lambda ratio for different intake pressures and engine speed at 2000 RPM.

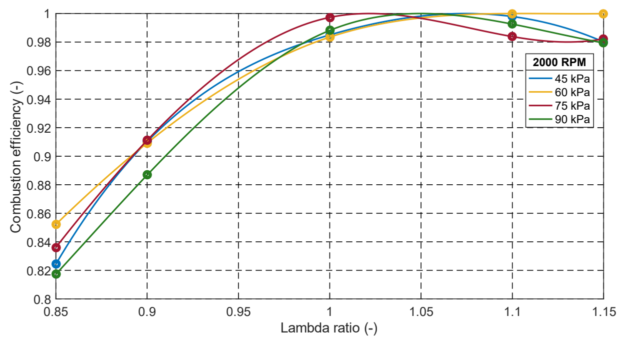


Figure 17: Normalized combustion efficiency as function of lambda ratio for different intake pressures and engine speed at 3000 RPM.

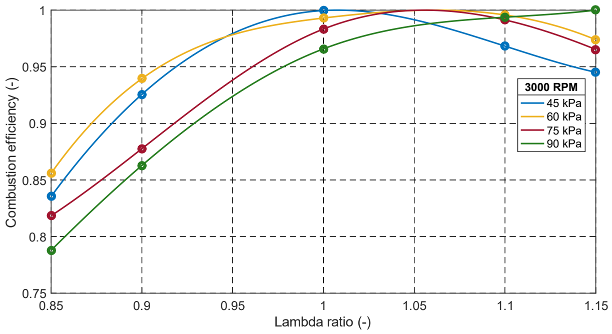
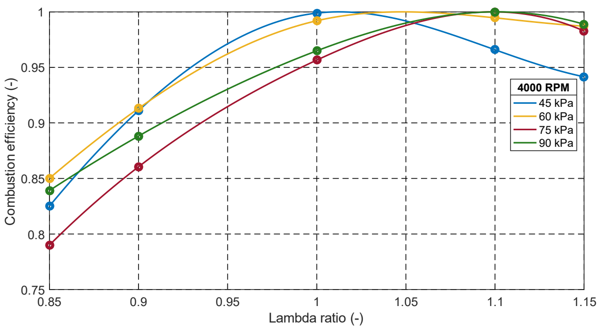


Figure 18: Normalized combustion efficiency as function of lambda ratio for different intake pressures and engine speed at 4000 RPM.



To explore the impact of the air-fuel ratio on the opening of the parabolas, referring to equation 8, tests were conducted at 2000 RPM for various intake pressures. The results obtained are shown in figures 19 to 22. The literature widely disregards the impacts of the air-fuel ratio on equation 8, given that the impact of the ignition timing on the efficiency factor $e_\zeta(\zeta)$ is normalized [1]. It can be observed from the results that, indeed, to simplify the model, this influence can be neglected.

Figure 19: Effective torque as function of ignition timing and lambda ratio for engine speed at 2000 RPM and intake pressure at 45 kPa.

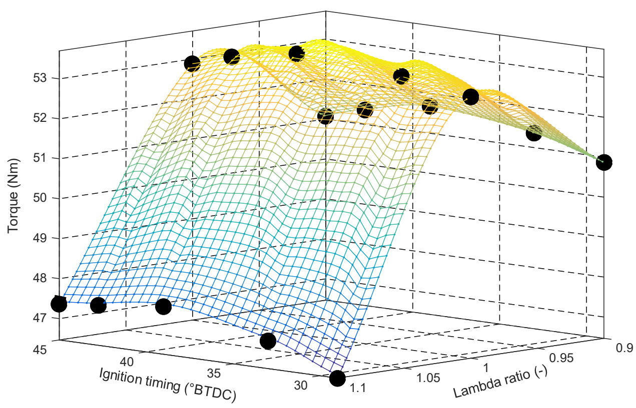


Figure 20: Effective torque as function of ignition timing and lambda ratio for engine speed at 2000 RPM and intake pressure at 60 kPa.

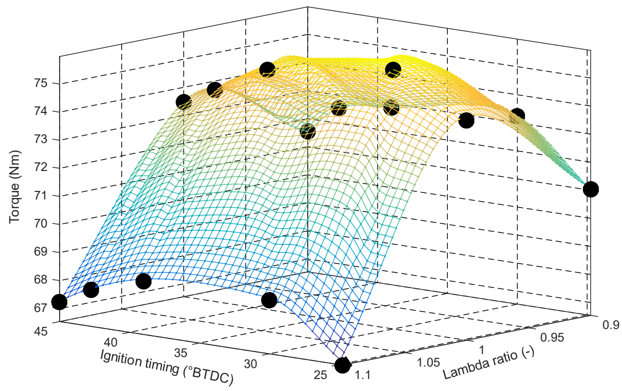


Figure 21: Effective torque as function of ignition timing and lambda ratio for engine speed at 2000 RPM and intake pressure at 75 kPa.

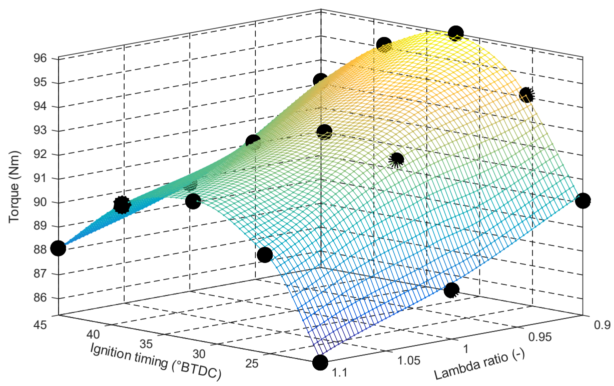
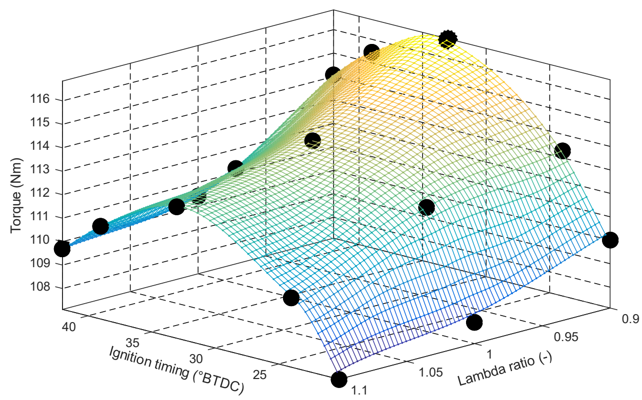


Figure 22: Effective torque as function of ignition timing and lambda ratio for engine speed at 2000 RPM and intake pressure at 90 kPa.



CONCLUSION

Based on the results obtained in this study, it is concluded that both the air-fuel ratio and the ignition timing play a crucial role in the engine performance, directly affecting the effective torque and indicated efficiency. It was observed that the relationship between torque and ignition timing follows a parabolic pattern, indicating that there exists an optimal ignition angle to achieve maximum torque. Furthermore, it was found that the air-fuel ratio has little interference in the parabolic relationship between torque and ignition angle, suggesting that other factors may be more influential in this relationship. However, the study revealed that the air-fuel ratio plays a significant role in effective torque when considering combustion efficiency. An improper air-fuel ratio can result in incomplete or inefficient combustion, leading to a decrease in effective torque.

In summary, this study contributes to a deeper understanding of the factors affecting effective torque and indicated efficiency in internal combustion engines. By considering the air-fuel ratio and ignition timing, it has been demonstrated that both play distinct but interconnected roles in engine performance. These findings have important implications for the design and optimization of engines, allowing for better utilization of available energy and improved combustion efficiency.

REFERENCES

[1] GUZZELLA, Lino; ONDER, Christopher. **Introduction to modeling and control of internal combustion engine systems**. 2. ed. Springer Science & Business Media, 2010.

[2] MAGNUSSON, Janek. **An investigation of maximum brake torque timing based on ionization current feedback.** 2007. Master's Thesis. Linkopings Universitet.

[3] BRUNETTI, Franco; GARCIA, Oswaldo. **Motores de combustão interna.** São Paulo: Blucher. v.1, 2012.

[4] NÜESCH, Sandro Patrick. **Analysis and control of multimode combustion switching sequence**. 2015. Tese de Doutorado. The University of Michigan.

[5] ERIKSSON, Lars; NIELSEN, Lars. **Modeling and control of engines and drivelines**. John Wiley & Sons, 2014.

[6] GERHARDT, Jurgen; HÖNNINGER, H.; BISCHOF, H. **A new approach to functional and software structure for engine management systems-Bosch ME7**. SAE technical paper, 1998.

[7] MOSKWA, John Joseph. **Automotive engine modeling for real time control**. 1988. Tese de Doutorado. Massachusetts Institute of Technology.